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The Effects of Oil Supply Pressure at Different Groove Position on Frictional Force and Torque in Journal Bearing Lubrication

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Abstract

An axial groove is a common supply method for distributing lubricant within a journal bearing. Lubricant is generally fed at a specific supply pressure to ensure that the journal and the bearing surface are separated. Shearing action between lubricant and bearing parts creates friction which contributes to power loss in the bearing. In this study, experimental work was conducted to determine the effect of oil supply pressure at different oil groove positions on torque and frictional force in hydrodynamic journal bearing. The journal bearing test rig with a journal diameter of 100mm and a length-to-diameter ratio of ½ was used. The oil supply pressure was set at 0.2, 0.5 and 0.7 MPa. The groove was positioned at 7 different locations of -45^0, -30^0, -15^0, 0^0, $+15^0$, $+30^0$ and $+45^0$. Measurements of frictional force, torque and friction coefficient were obtained for speed values of 500 and 800 rpm at 10 and 15 kN radial loads. It was found that oil supply pressure and groove positions had affected the frictional force and torque in journal bearing.

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Keywords: Hydrodynamic Lubrication; Oil inlet pressure effects; Oil groove position effects; Friction Coefficient

1. Introduction

A journal bearing is common in many types of machinery. The main function is to support radial load and facilitate motion as well as transfer of power. A journal bearing consists of two main components where the shaft called journal rotates freely in its bushing. Basu et.al [1] identified a journal bearing as an important member of hydrodynamic slider family with the simplest construction. The bearing operates with a small clearance of the order of 1/1000 of the journal radius. The clearance is filled with lubricant and this lubricant layer is responsible for the load carrying capacity of the journal bearing. The lubrication mechanism in a journal bearing is often referred to as hydrodynamic lubrication which has been widely studied and reported. Hydrodynamic fluid film lubrication occurs when the journal and bearing surfaces are totally separated by the established fluid layer. In the case of journal bearing, the oil film needs to be thick enough to prevent asperity contact. During the development or design stage of a journal bearing, several important parameters or characteristics in hydrodynamic lubrication such as film thickness, viscosity, pressure, temperature and friction are considered to get optimum operations.

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In previous studies [2-5], investigations on viscosity, film thickness, temperature and pressure related to journal bearings were described. There are also studies related to oil supply pressure which specifically on thermal affects in journal bearings [6-8]. Computational studies by [9, 10] using numerical method have investigated the influence of oil supply pressure and its position for long journal bearings. They found that oil supply pressure does affect static and dynamic performances of journal bearing. Authors latest studies, [11, 12] found that oil supply pressure and oil groove position do affect pressure profiles, torque and frictional force in journal bearing. In this study, the effect of oil supply pressure and groove positions on fluids friction and torque in hydrodynamic journal bearing has been investigated.

1.1. Oil Groove Supply

In hydrodynamic analysis, the oil supply was assumed available to flow into the bearing at least as fast as it leaks out. In this study, the oil was fed to the system by an oil supply hole and groove. Lubricant lost due to a side leakage must be compensated to prevent depletion of lubricant inside the bearing. Oil enters the groove through an oil supply hole and flows either by gravity or under pressure. Ideally, the groove should be the same length as the bearing but this would cause all lubricant to leak from the sides of the groove [13]. In this experimental study, a short angle groove type was used. Lubricant oil supplied to the bearing was pressurized to reduce lubricant heating, viscosity loss, prevent shaft to bush contact during starting and stopping, modify vibration stability and suppressed cavitation. Suitable oil supply flow can be calculated by Couette flow. Costa et.al [8] reported that increasing oil supply pressure can reduce the operating temperature and increase the maximum circumferential hydrodynamic pressure. This is consistent with findings by [6] which concluded that the oil supply pressure and the geometry of the feed control determine the cooling effects.

1.2. Friction in a Journal bearing

In sliding phenomenon, lubrication helps to reduce friction. Small frictional forces will result in the dissipation of energy and consequent loss of machine efficiency. The friction value in the journal bearing can be calculated by integrating the shear stress over the bearing area. For the Half-Sommerfeld condition, friction in journal bearing is given by [13],

$$F = \frac{2\pi\eta ULR}{c}\frac{1}{(1-\varepsilon^2)^{0/5}} \tag{1}$$

Where, η is viscosity (Pa.s), U is rotational speed (rps), L is bearing length (m), R is journal radius (m), $\boldsymbol{c}$ is clearance (m) and ε is eccentricity ratio. This equation reduces to the simplified expression for Petroff friction when the shaft and bush are concentric. There are three important assumptions in Eq. 1: (i) Bearing surfaces are smooth, (ii) The fluid is Newtonian and the flow is laminar, and (iii) Inertia force resulting from acceleration of the fluid and body forces are small compared with the surface force.

2. Apparatus and Experimental Procedure

2.1. Journal Bearing Test Rig

The Journal Bearing test rig in Fig. 1(a) was used in this experiment. The loading arm is mounted to the bearing. The frictional force sensor is mounted on the spindle housing as shown in Fig. 1(b). When the loading arm presses the loading pin, the force sensor will record the friction values. A pneumatic bellow is used to apply the required load. The maximum speed of the journal test rig is 1000 rpm. The speed values used for testing were 500 and 800 rpm.

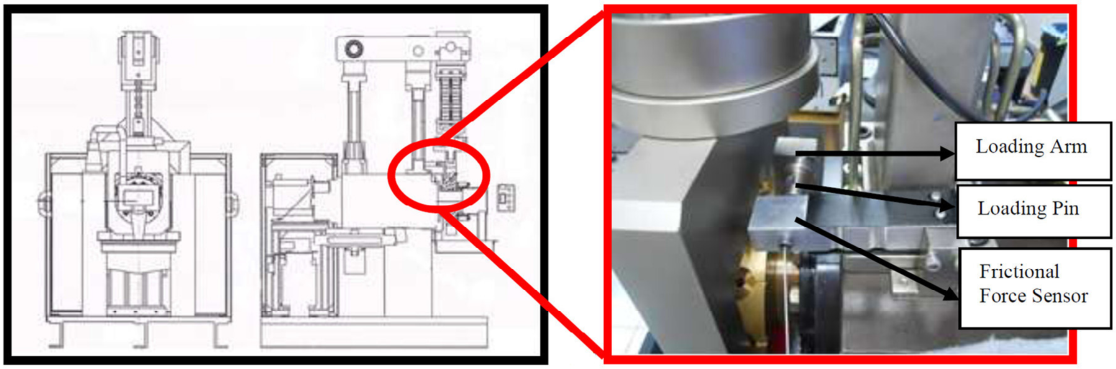

Fig. 1. (a) Journal bearing test rig diagram and (b) Pressure sensors attached to the bearing system.

The tests were conducted at different loads (10, 15 kN). Oil pressure supply was set at 0.2, 0.5 and 0.7 MPa. The groove position was varied at -45^0, -30^0, -15^0, 0^0, +15^0, +30^0 and +45^0 from the loading arm. Details of test bearing dimensions, lubricant properties and operating parameters are given in Table 1.

Table 1. Bearing Dimensions, lubricant properties, operating parameters and sensor specifications

Parameter	Values
Journal diameter, D	100 mm
Bearing Length, L	50 mm
Radial clearance, c	52 µm
Load range, W	10, 15 kN
Journal Speed	500 – 800 RPM
Lubricant	Shell Telus 68
Lubricant viscosity	68 cSt @ 40oC
	8.8 cSt @ 100oC
Load sensor	
Model	Sensortronic
Range	30kg
Type	Beam
Accuracy	0.01± 1% Nm

3. Result and Discussion

Results of frictional force, torque and friction coefficient obtained were plotted as shown in Fig. 2 to Fig. 6.

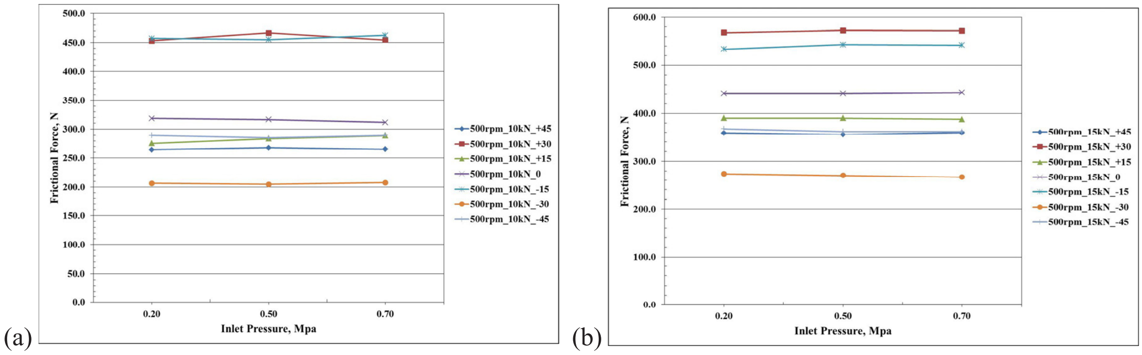

Fig. 2. (a) Frictional force at 500 rpm of 10 kN; (b) Frictional Force at 500 rpm of 15 kN for different oil inlet pressures and groove locations.

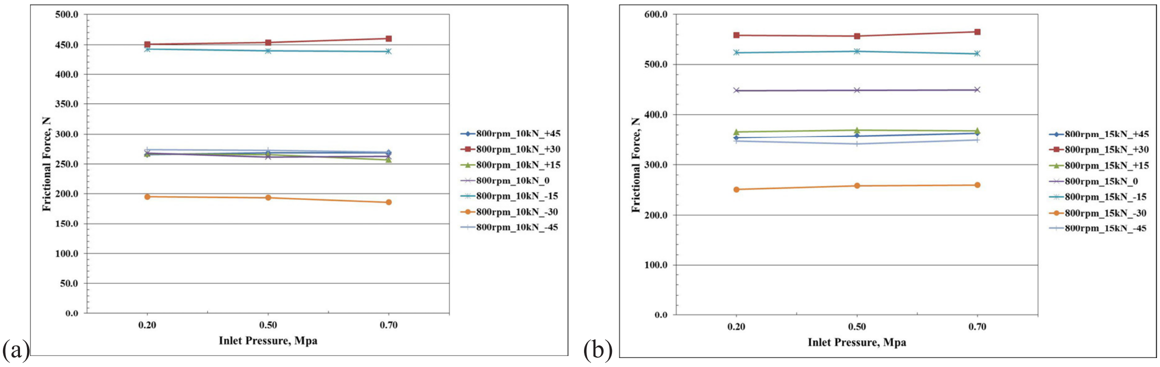

Fig. 3. (a) Frictional force at 800 rpm of 10 kN; (b) Frictional Force at 800 rpm of 15 kN for different oil inlet pressure and groove locations.

Fig. 2(a) shows bearing frictional forces obtained for 500rpm speed at 10 kN load, while Fig. 2(b) for 15 kN load. In both figures, it was observed that the groove position of -30^0 has a lower frictional force value as compared to other positions. Increasing of oil inlet pressure at this groove position tends to increase the frictional force for load of 10 kN. On the contrary, the frictional force value has decreased for the case of 15 kN load. Increasing the bearing speed to 800 rpm, as shown in Fig. 3(a) for load of 10 kN and Fig 3(b) for load of 15 kN, has caused a decrease in bearing frictional force. This can be observed for groove position of -30^0 in both figures.

Frictional forces obtained were converted to bearing torque as shown in Fig. 4 and Fig. 5. Bearing torque for speed of 500rpm at loads of 10 kN and 15 kN are shown in Fig. 4(a) and Fig. 4(b) respectively. In the case of 800rpm, the bearing torque results obtained for 10kN and 15 kN are given in Fig. 5(a) and 5(b) respectively. All figures were plotted for different groove positions and different oil inlet pressure values.

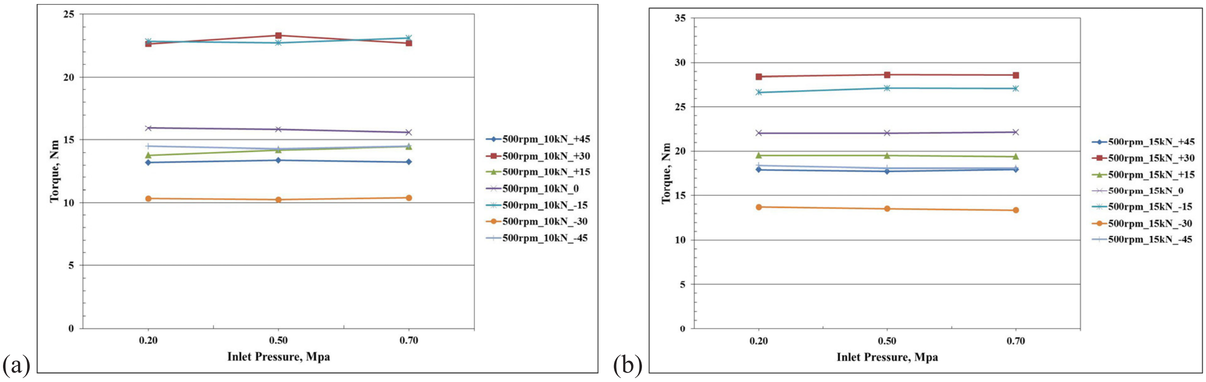

Fig. 4. (a) Bearing torque at 500 rpm of 10 kN; (b) Bearing torque at 500 rpm of 15 kN for different oil inlet pressures and groove locations.

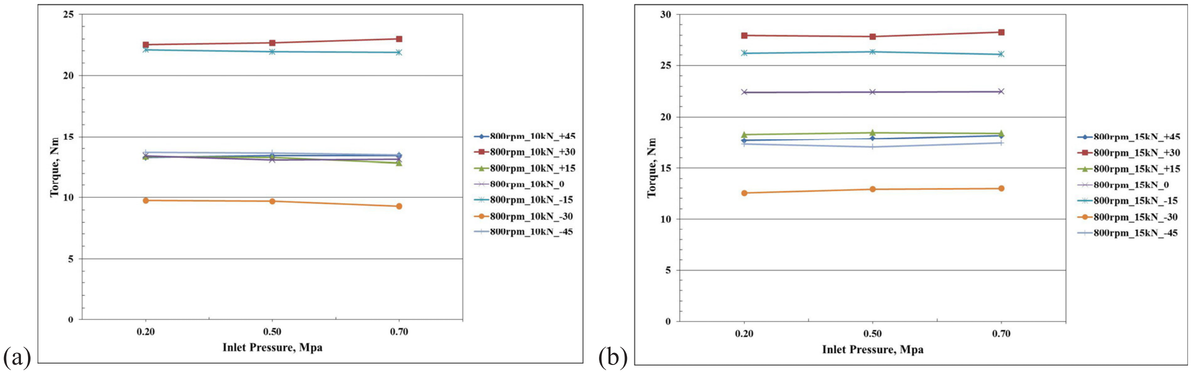

Fig. 5. (a) Bearing torque at 800 rpm and 10 kN; (b) Bearing torque at 800 rpm and 15 kN for different oil inlet pressures and groove locations.

Theoretically, from Eq. (1), changing of groove location and oil pressure supply should not affect the frictional force. However, from experimental results obtained it is clearly shown that frictional force was affected by the groove position and the oil supply pressure to some extent.

Fluid friction coefficient were calculated and plotted in Fig. 6 for speed of 500 and 800 rpm at loads of 10 and 15 kN. Groove location of -30^0 shows the lowest value of fluid friction coefficient compared to other groove locations. Increasing oil inlet pressure at this groove position has decreased fluid friction coefficient as show in Fig. 6(a). This finding requires further investigation which is not described in this paper. In both speed values, groove position of 30^0 gives the highest value in friction coefficient. In some cases such as at speed of 500 rpm at 10 kN load for groove location of 15^0, increasing oil inlet supply pressure (from 0.2 to 0.5 to 0.7 MPa) has increased friction coefficient obtained. In other cases, a different trend was observed where an increase in friction coefficient was observed initially as the oil inlet supply pressure was regulated from 0.2 to 0.5 MPa but started to decrease when the pressure went from 0.5 to 0.7 to MPa.

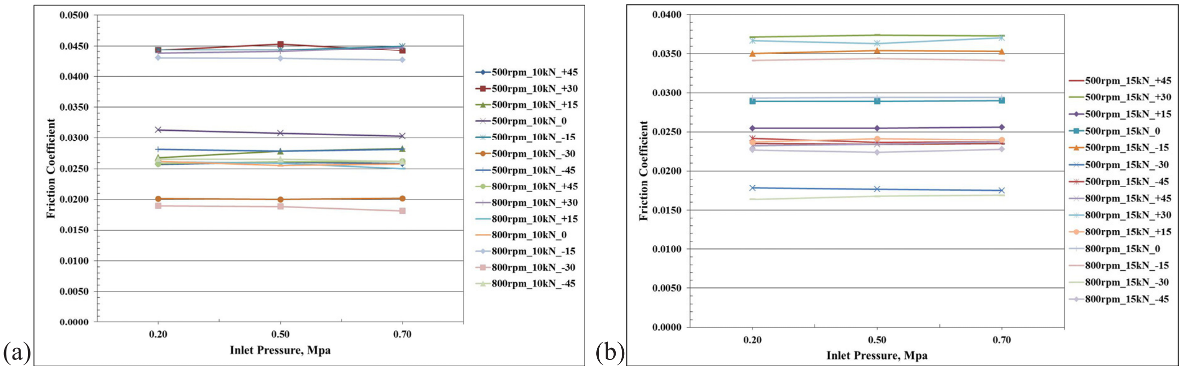

Fig. 6. Friction Coefficient for 500 and 800 rpm at (a) 10 kN and (b) 15 kN for different oil inlet pressures and groove locations

4. Conclusion

The effects of oil supply pressure on torque, frictional force and friction coefficient in hydrodynamic journal bearing at different groove locations have been studied. Experimental results have been presented for different speeds (500 and 800rpm) and loads (10 and 15 kN) when the oil supply pressure and groove location were varied. The following conclusions can be drawn from this experiment:-

- Torque and frictional force tend to change when the groove position is changed. At certain positions increasing oil supply pressure will increase torque and frictional force of the bearing.
- Fluid friction coefficient of 15 kN load is higher compared to that of 10 kN. The groove position of -30^0 has a lower friction coefficient value for speed values of 500 and 800 rpm.

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